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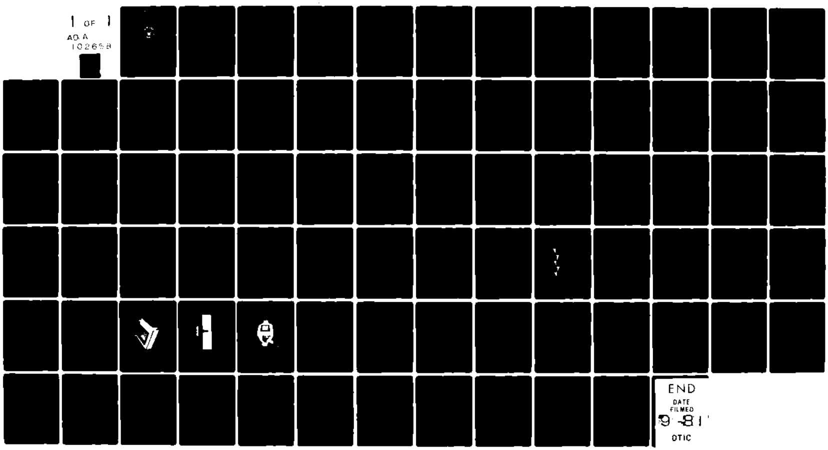

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THESIS

A TEST CONDENSER TO MEASURE CONDENSATE INUNDATION EFFECTS IN A TUBE BUNDLE

by

Roger Harold Morrison

March 1981

Thesis Advisor: Dr. Paul J. Marto

Approved for public release; distribution unlimited

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Validation of system operation was accomplished using steam at 253 mm Hg absolute. Recommendations to improve the test condenser are provided.

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A Test Condenser to Measure Condensate Inundation Effects in a Tube Bundle

by

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Submitted in partial fulfillment of the requirements for the degree of

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ABSTRACT

A test condenser was designed and constructed to provide a means to evaluate the effects of condensate inundation on enhanced condenser tubing. Fifteen horizontal tubes were used to form a square inline tube matrix three tubes across and five tubes high with a spacing to diameter ratio of 1.5. The center column of active tubes was made of 16 mm O.D. half hard copper. They were flanked by full round 16 mm O.D. stainless steel dummy tubes.

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Validation of system operation was successfully accomplished using steam at 253 mm Hg absolute. Recommendations to improve the test condenser are provided.

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NOMENCLATURE

A - Heat transfer area of one tube (m^2)

A_c - Cross sectional area of the condenser (m^2)

C - Specific heat at constant pressure ($kJ/kg\text{-}^oK$)

C_1 - Amount of condensate actually condensed by one tube per hour (kg/hr)

C_N - Withers, Young, and Lampert [Ref. 22] correction to Nusselt multiple tube equation

D - Diameter (m)

D_o - Outside diameter (m)

F_d - Eissenberg [Ref. 11] tube bank parameter

g - Acceleration of gravity ($9.81\ m/sec^2$)

G_i - Amount of condensate formed per hour on the $i^{\underline{th}}$ tube (kg/hr)

G_N - Amount of condensate formed per hour by the $N^{\underline{th}}$ tube (kg/hr)

h - Heat transfer coefficient ($W/m^2\text{-}^oK$)

h_{fg} - Latent heat of vaporization (kJ/kg)

h_N - Local heat transfer coefficient for the $N^{\underline{th}}$ tube in a column ($W/m^2\text{-}^oK$)

$\bar{h}_N$ - Average heat transfer coefficient for a column of N tubes ($W/m^2\text{-}^oK$)

h_{Nu} - Heat transfer coefficient calculated from the Nusselt equation ($W/m^2\text{-}^oK$)

h_1 - Experimentally determined value for the heat transfer coefficient of the first tube in a bundle ($W/m^2\text{-}^oK$)

k - Thermal conductivity ($W/m\text{-}^oK$)

$\dot{m}_{cond}$	- Total mass flow rate of the condensate (kg/sec)
N	- Number of tubes in a column
Pr	- Prandtl number
Q	- Heat transferred per unit time (W)
r	- Radius (m)
S/D	- Spacing to diameter ratio
T_{ci}	- Cooling water inlet temperature
T_{co}	- Cooling water outlet temperature
T_s	- Saturation temperature of steam (^oC)
T_w	- Tube wall surface temperature (^oC)
v	- Specific volume of the vapor at the test section (m^3/kg)
w	- Amount of condensate formed on the $N^{\underline{th}}$ tube per unit time (kg/hr)
W	- Amount of condensate draining onto the $N^{\underline{th}}$ tube per unit time (kg/hr)
α	- Thermal diffusivity (m^2/hr)
ΔT	- Temperature difference (^oC)
ξ	- Acceleration parameter ($k\Delta T/\mu h_{fg}$)
ν	- Kinematic viscosity (m^2/hr)
μ	- Dynamic viscosity (kg/m-s)
ζ	- Heat capacity parameter ($C\Delta T/h_{fg}$)
ρ	- Density (kg/m^3)
ρ_v	- Vapor density (kg/m^3)

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I. HISTORICAL BACKGROUND

The Navy is interested in smaller and lighter high performance condensers for two reasons. First, advanced condensers reduce the total weight of a steam propulsion plant by reducing the size and amount of material required. Depending upon the condenser material, initial and replacement costs could be reduced. The steam cycle efficiency is increased and operating costs are reduced because of the superior heat transfer capability. Secondly, combat effectiveness is increased because the cruising radius is increased due to increased fuel storage (due to the reduced volume and weight of the condenser) and the better fuel economy.

Work in this area was begun at the Naval Postgraduate School by H. T. Search [Ref. 1] in 1977 who conducted an extensive feasibility study in which he states that naval condensers are significantly overdesigned. He estimated that a fifty percent increase in heat load for constant pumping power could be achieved using enhanced heat transfer techniques. Alternatively, enhancement techniques could reduce the weight and volume of naval condensers by as much as forty percent with constant heat load but at the expense of pumping power. He recommended that a systematic study of enhancement techniques be done.

The research done at the Naval Postgraduate School into heat transfer enhancement of a single tubes was done by Beck [Ref. 2], Pence [Ref. 3], Reilly [Ref. 4], Fenner [Ref. 5], and Ciftci [Ref. 6]. As an example of the results obtained, Ciftci found that the overall heat transfer coefficient (corrected to remove the wall resistance) of a 1 inch O.D. enhanced tube may be as much as 1.9 times that of a smooth tube of the same diameter.

In a separate paper, Marto, Reilly, and Fenner [Ref. 7] report the following conclusions after investigating the improvement in the overall heat transfer coefficient of an enhanced single tube in a dummy tube nest:

1) The enhanced tubes showed an improvement in corrected overall heat transfer coefficient of as much as 104 percent over that of a smooth tube.
2) Most of the improvement occurred on the cooling water side of the tube presumably due to a combination of increased surface area as well as increased turbulence and swirl in the tube flow.
3) The increase in heat transfer occurred at the expense of a substantial increase in cooling water pressure drop with friction factors as much as ten times that of smooth tubes.
4) Enhanced tubes manufactured with very deep grooves exhibited the best heat transfer performance, but vibrations due to cooling water turbulence could occur if the tubes were not properly supported.

5) For a given tube geometry, at a constant groove depth, heat transfer performance was dependent upon groove pitch or helix angle.

6) The use of enhanced tubes in a surface condenser may reduce the required surface area by as much as fifty percent.

All of these researchers concluded that vapor velocity and condensate inundation effects must be included in the study of enhanced tubes before the performance of enhanced tubes could be fully understood.

Eshleman [Ref. 8] converted the experimental apparatus originally designed by Beck for single tube experiments to accept a column of five active horizontal tubes so that the effects of condensate inundation could be studied. Dummy half tubes were placed on either side of the active tubes to provide a vapor flow path. A staggered arrangement was chosen with a spacing-to-diameter ratio of 1.5. Utilizing smooth tubes, he found that the data for tubes 1 to 3 correlated with the Nusselt prediction, but that tubes 4 and 5 were well below this prediction. The drop in performance of the last two tubes was attributed to either a low mass flow rate of steam remaining to be condensed or a concentration of noncondensable gases in the lower part of the test section. The presence of noncondensable gases was the chief suspect as the values for tubes 4 and 5 were obtained after long periods of test apparatus operation. The noncondensable gases had several means of entering the system such as with the house steam, desuperheater spray, or leaks in the installed piping.

Eshleman also observed lateral droplet migration from the ends toward the centers of the tubes. This migration was probably due to secondary vapor flow in the test section, which causes a local thickening of the film thickness at the center of the tube reducing its heat transfer performance. The secondary flow could also create pockets where noncondensable gases could be trapped, further reducing tube performance.

Demirel [Ref. 9], after modifying Eshleman's test rig, found that the performance of the first three tubes (again smooth) was slightly above Nusselt's prediction. The values for the last two tubes took a dramatic jump upwards and were considerably higher than those obtained for the first three tubes. The cause for this was thought to be secondary flow within the condenser as droplet migration was again observed.

The work in this area by Eshleman and Demirel was inconclusive because of the discrepancy in their data for the last two tubes of the five tube column. The objectives of this thesis were to resolve this discrepancy in tube performance by:

1) designing a new test section to eliminate the droplet migration phenomenon; and
2) obtaining baseline performance of the test section utilizing smooth tubes.

II. THEORETICAL BACKGROUND

A. PREDICTIONS OF TUBE BUNDLE PERFORMANCE

The foundation for most of the present day knowledge concerning condensation on plates and horizontal tubes is based on work first done by Nusselt circa 1916. Nobbs [Ref. 10] notes that Nusselt's work included the following assumptions for single tubes:

1) the wall temperature is constant;
2) the flow is laminar in the condensate film;
3) heat transfer in the condensate is by conduction perpendicular to the condensate surface only and subcooling may be neglected;
4) fluid properties are constant within the condensate layer;
5) the forces due to hydrostatic pressure, surface tension, inertia and vapor-liquid interfacial shear are negligible when compared to the viscous and gravitational forces;
6) the surrounding steam and vapor/liquid interface are at saturation temperature;
7) the film thickness is small when compared with normal tube diameters and the effects of curvature are small.

Eissenberg [Ref. 11] notes that two additional assumptions are required for Nusselt conditions when considering a column of horizontal tubes:

8) condensate drains as a laminar sheet from a tube bottom to a tube top such that velocity and temperature gradients are not lost in the fall between tubes;

9) the saturation temperature and the tube wall temperature are constant for all tubes in the bank.

Returning to Nobbs' discussion, he states that for a single horizontal tube, Nusselt derived the following equation:

$$h_{Nu} = .728 \left[\frac{k^3 \rho \; (\rho - \rho_v) h_{fg} \; g}{\mu \; D \; (T_s - T_w)}\right]^{1/4} \qquad (1)$$

This equation was found to be satisfactory when experiments were conducted that closely approximated the Nusselt condition.

Jakob [Ref. 12] modified the single tube equation of Nusselt to arrive at the following equation for a tube column:

$$\overline{h_N} = .725 \left[\frac{k^3 \rho \; (\rho - \rho_v) h_{fg} g}{\mu N \; D \; (T_s - T_w)}\right]^{1/4} \qquad (2)$$

where $\overline{h}_N$ is the average heat transfer coefficient for the column and N is the number of tubes in the column. Nobbs states that this equation is too pessimistic, most likely due to the effects of rippling, splashing, and turbulent flow of the condensate. These effects all serve to improve the heat transfer characteristics of the tube by changing the film thickness.

The usual expression seen for the Nusselt result is the value of the heat transfer coefficient normalized by the value of the first tube:

$$\frac{\overline{h_N}}{\overline{h_{Nu}}} = N^{-1/4} \qquad (3)$$

Eissenberg [Ref. 11] notes that another normalized result within the Nusselt framework is:

$$\frac{h_N}{\overline{h_{Nu}}} = N^{3/4} - (N-1)^{3/4} \qquad (4)$$

where h_N is the value for the heat transfer coefficient of the $N^{\underline{th}}$ tube.

According to Nobbs, Fuks [Ref. 13] experimentally determined an equation for the heat transfer coefficients of a tube bank. Fuks used a staggered tube bundle of 19mm OD tubes with steam flowing vertically downwards. He examined the effects of condensate drainage by obtaining data down to the eleventh row of the bundle. This data was supplemented by information from tests utilizing data obtained from a flooding tube supplying saturated water to the top of the bundle. Fuks' correlation, published in 1957, is in the following form (where the effect of vapor velocity has been eliminated):

$$\frac{h_N}{\overline{h_1}} = \left[\frac{W + w}{w}\right]^{-0.07} \qquad (5)$$

where W is the condensate draining onto the $N^{\underline{th}}$ tube per unit time, w is the condensate formed on the $N^{\underline{th}}$ tube per unit time, and h_N is the heat transfer coefficient of the $N^{\underline{th}}$ tube.

Borishankii and Paleev [Ref. 14] in 1964 note that Fuks obtained the equation in the following forms:

$$\frac{h_N}{h_1} = \left[\frac{W + w}{w}\right]^{-0.07} = N^{-0.07} \tag{6}$$

Isachenko and Gluskov [Ref. 15] in 1969 were able to verify Fuks' equation. Their first series of tests put the working tube under the overflow tube at a distance of 20mm between centers (corresponding to a relative pitch of S/D = 1.67, an inline arrangement). The second series of tests used a staggered arrangement. Their data very nearly matches that of Fuks' equation. Nobbs [Ref. 10] notes that a better fit of their data would be given by an exponent of -0.125 instead of -0.07. Nobbs also notes that Turek [Ref. 16] obtained agreement with Fuks' model for tube columns of up to eight tubes. Beyond a column of eight tubes, the experimental values obtained were below those of Fuks.

In 1961, Chen [Ref. 17] derived a formula to predict the heat transfer coefficient of a tube bundle utilizing boundary conditions that were influenced by the following effects:

1) the momentum gain of the falling condensate between tubes, and

2) the condensation of vapor on the condensate between tubes.

The result of this derivation is

$$\frac{\overline{h_N}}{\overline{h_{Nu}}} = N^{-1/4} \left[1 + 0.2\, \zeta\, (N-1)\right] \cdot$$

$$\left[\frac{1 - 0.68\, \zeta + 0.02\, \zeta\, \xi}{1 + 0.95\, \xi - 0.15\, \zeta\, \xi}\right]^{1/4} \tag{7}$$

where $\xi = k\Delta T/\mu h_{fg}$ (the acceleration parameter) and $\zeta = C\Delta T/h_{fg}$ (the heat capacity parameter). The two parameters are related by $\xi = \zeta/Pr$.

Eissenberg [Ref. 11] in 1972, recognizing the interaction of steam velocity, inundation and noncondensable gases on the heat transfer coefficient, designed an experiment which investigated these parameters. His experimental design utilized five instrumented active tubes that were surrounded by twenty seven identical active tubes. These active tubes were surrounded upstream and downstream by dummy tubes to provide entrance and exit flow conditions for the active array. The five intrumented active tubes were arranged vertically within a bundle of tubes arranged in a staggered triangular array. Steam flow was horizontal.

As a result of his experiments, Eissenberg developed a side drainage model. In this mode, condensate may drain to tubes that are not directly below one another (refer to Fig. 1); in doing so, Nusselt's assumption concerning condensate drainage has been violated. Side drainage may depend on the interaction of the following:

1) Orientation - triangular spaced patterns are more susceptible to side drainage than inline tube bundles;
2) Spacing - the smaller S/D becomes, the more frequently side drainage should occur;
3) Momentum - the greater the horizontal component of momentum of a drop leaving the tube, the greater the side drainage;
4) Steam velocity - when steam flows horizontally across tubes at sufficient velocity, the drop trajectory will reflect the added lateral momentum. When steam flows vertically, its direction change with each tube may also impart lateral momentum to the drops; and
5) Misaligned tubes - a tube misaligned in a bundle may receive greater or lesser amounts of condensate depending on its orientation with respect to the side tubes.

If the condensate drains purely by the side drainage model, Eissenberg predicted:

$$\frac{\overline{h_N}}{\overline{h_{Nu}}} = 0.60 + 0.42\ N^{-1/4} \qquad (8)$$

Eissenberg's experimental values fall between the above equation and Nusselt's prediction, Eq. (3).

Considering all factors such as drainage, steam velocity, noncondensable gases, etc., Eissenberg recommends the following approximate formula for the prediction of heat transfer coefficients:

$$\frac{\bar{h}_N}{\bar{h}_1} = 0.6\ F_d + (1 - .058\ F_d)\ N^{-1/4} \qquad (9)$$

where F_d is the tube bundle parameter. (Example: In a triangular layout with S/D of 1.33 and a diameter of 1 inch, F_d will have a value of 0.8. If S/D is greater than 2, F_d will have a value of zero.)

In 1975, Nobbs [Ref. 10] did experimental work for his doctoral dissertation to investigate the combined effects of downward vapor velocity and condensate inundation on the condensate rates in horizontal tube banks. Nobbs utilized two tube bank configurations, one a square in-line arrangement and the other a staggered equilateral triangular arrangement. Each consisted of three complete columns with half columns on the walls, and there were six and seven rows in the in-line and triangular arrangements respectively. The pitch-to-diameter ratio was 1.25 with a nominal tube ID of 19.05 mm. Due to data scatter, he was unable to find a general correlation, but the data fell between extremes found by Fuks (Eq. 5) and Nusselt (Eq. 5). Nobbs supplied the following conclusions:

1) Vapor velocity increases the condensate heat transfer coefficient on both inundated and uninundated tubes in a tube bank;

2) The effect of inundation is generally to reduce the heat transfer coefficient. The rate of reduction with increase in inundation rate becomes smaller as the vapor velocity is increased; and

3) The condensate drainage path, particularly in a triangularly shaped tube bank, is often not vertically downwards but in a diagonal direction. This can lead to some tubes receiving no inundation, and others receiving more than their proportionate share.

Eissenberg [Ref. 11] notes that experimental results differ from each other by significant amounts as well as deviating from Nusselt theory. These disagreements have not been the subject of theoretical analysis. Instead, the experimental results are fitted, in most cases, to an equation using the following form:

$$\frac{\overline{h_N}}{\overline{h_{Nu}}} = N^{-S} \qquad (10)$$

where the exponent S is empirically determined and N is the number of tubes in the column. Values of S in the range 0.07 to 0.20 have been reported.

Davidson and Rowe [Ref. 18] suggest a prediction that is based on:

$$\frac{h_N}{\bar{h}_1} = [\frac{W}{C}]^{-\gamma} \qquad (11)$$

where W is the total amount of condensate leaving the N^{th} tube and C is the amount of condensate actually condensed on the tube. The exponent, γ, is based on the mean steam flow rate with the variation in γ being reported from 0.07 to 0.223.

Figure 2 is a comparison of the predictions of Nusselt (Eq. 3), Eissenberg (Eq. 8), and Chen (Eq. 7) for the normalized average value of the heat transfer coefficient of a bank of N tubes. Figure 3 is a comparison of the normalized predictions for the heat transfer coefficient of the N^{th} tube in a tube bank.

B. INVESTIGATION INTO THE MECHANISM OF INUNDATION

One of the main areas of contention concerning the Nusselt assumptions has been the actual process of inundation. There has been a great deal of research into this area, and a short review of the literature follows.

Kutateladze's paper [Ref. 19], published in 1952, notes that Nusselt assumed a continuous flow of condensate in a sheet while the flow of condensate actually takes place periodically, in separate drops. Kutateladze states that this does not affect the heat exchange for the single tube because even in this case, the flow of condensate is still symmetrical with respect to the vertical axis. The heat transfer coefficient represents some average with time. Consequently, there is a redistribution of condensate because of surface tension which tends to maintain the free surface at a minimum. The dimensions of the drops separating from the wall are determined by the interaction of the force of gravity and that of surface tension.

The dimensions of the drops become smaller with an increase in vapor pressure because of a decrease in the surface tension of the condensate.

For tube bundles, two additional factors must be considered:

1) condensate from tubes above runs down on each tube below;

2) the film profile is not symmetrical because the drops from above do not fall exactly on the upper part of the tube.

As a result, the Nusselt number for any tube in the absence of vapor velocity, may be expressed as:

$$Nu_D = \frac{hD}{k} = f\left(\sum_{i=1}^{N} \frac{G_i}{G_N}\right) \left(\frac{\nu}{\alpha} \frac{gD^3}{\nu^2} \frac{h_{fg}}{C\Delta T}\right)^{1/4} \tag{12}$$

where G_i is the amount of condensate formed per hour on the tube under consideration, and G_N is the amount of condensate condensed on tube N.

Kutateladze's experimental results based upon spacing-to-diameter ratios of 1.5 to 3 revealed that the effect of vertical distance between tubes on heat transfer is almost nil. This dimension has no effect on the distribution of condensate along the tube. An increase in this dimension only increases the force of the drops falling from above onto the tube in question which does not affect the heat transfer for the usual (S/D = 1.25 - 2.00) spacing-to-diameter ratios.

In a paper published in 1969, Borishankii and Paleev [Ref. 14] state that the effect of descending liquid flow on local and average heat transfer coefficients is much smaller than the effect predicted by the theoretical calculations based on the Nusselt scheme of continuous downward flow. For any fairly large tube bundle, except for a single row of tubes in a vertically unlimited space, the vapor cannot be regarded as stagnant since each row of tubes is washed by the vapor stream flowing to the subsequent rows. The vapor velocity is greatest at the first row of tubes and is determined here by the amount of vapor condensing on all the tubes in the bank. The condensate does not flow from each tube in a continuous film. Rather it forms drops and films that separate at fixed intervals at relatively great distances from each other. In reality, the rate of heat transfer during vapor condensation on a bank of horizontal tubes is greatly affected by initial vapor velocity and by its change along the vapor path through the tube bank.

Shklover and Buevich [Ref. 20] in a 1978 paper, note that Nusselt's prediction of progressively falling heat transfer coefficients is qualitatively correct but much too pessimistic. Splashing, rippled films, and inundation in discrete drops all conspire to moderate the reduction expected due to theory. Nusselt assumed that the condensate runs as a continuous film off the lower generatrix of a tube. In fact, condensate is discretely removed from tubes in individual droplets and streams.

Shklover and Buevich analyzed the condensation process using movie films taken at rates of 24, 500 and 1000 frames-per-second. Analysis of these data showed that condensate dropping onto the tube situated directly below hardly spreads along the length of the tube, but rolls over its perimeter, locally thickening the film. Over the entire time of condensation in the bundles, the removal of condensate takes place in the form of a cluster of droplets which simultaneously comes into contact with the upper and lower tubes.

Yung, Lorenz, and Ganic [Ref. 21] in a 1980 study of vapor/liquid interaction in falling film evaporators make the following observations which are also applicable to condensers. A horizontal tube design is more vulnerable to vapor/liquid interaction. In the absence of vapor crossflow, unevaporated fluid from any given tube will fall directly on the next lower tube. At relatively low flow rates, the liquid falls in the form of droplets. At relatively high feed rates, the liquid falls in the form of columns. At still higher feed rates, the liquid falls as unstable sheets and columns. It must be noted that this last condition is not of interest when studying falling film evaporators. In a horizontal tube evaporator, the physical form of the liquid falling from one tube to the next depends on the liquid flow and the distance between tubes.

When the flow rate is small and the tube spacing large, the liquid is usually in the form of droplets which are generated at discrete points along the underside of the horizontal tube. As the drop detaches from the film, a long narrow tail is formed which, by the well-known Rayleigh instability, eventually breaks up into four or five droplets. In cases where the liquid is slightly deflected from the center of the tube, part of the flow may still climb up and wet the other side of the tube. For such a case, the averaged heat transfer coefficient is probably not significantly different from the case of evenly divided flow. If the drop is significantly deflected due to a crossflow velocity component, the overall heat transfer performance is not necessarily reduced because the liquid may experience a good "hit" on a tube in the adjacent column.

Splashing also affects the heat transfer coefficient. This occurs when liquid impinges on a solid surface which entrained by the vapor due to the "splash".

Yung, Lorenz, and Ganic concluded that good wetting of a tube, which is essential to good heat transfer performance, can be better achieved by a well controlled liquid flow rather than relying on a random hist from one tube to the next.

C. RESEARCH INTO ENHANCED TUBES

Studies on the heat transfer characteristics of single enhanced tubes are numerous. However, studies dealing with enhanced tubes in bundles are relatively few. A brief survey of two of these studies is presented here.

Withers, Young, and Lampert [Ref. 23] in 1975 described an experiment in which they investigated the heat transfer characteristics of corrugated (enhanced) tubes. The experimental apparatus consisted of tubes placed horizontally in a triangularly pitched layout. A separate supply of distilled water was circulated in one pass through three rows of tubes. Control instrumentation permitted round-the-clock operation of the steam condensing system.

Their work resulted in an equation which produces an expression for the average heat transfer coefficient of a bank of N tubes in the following form

$$\overline{h_N} = 0.725\ C_N \left[\frac{k^3\ \rho^2\ g h_{fg}}{\mu N\ D_o\ \Delta T_f}\right]^{1/4} \qquad (13)$$

where C_N is the correction factor to the Nusselt multiple tube equation. They found that C_N was a function of N which could be expressed in the form $C_N = A(N)^b$. If the equation is normalized with respect to the Nusselt value for the first tube, the relation becomes

$$\frac{\overline{h_N}}{\overline{h_{N_u}}} = C_N\ N^{-1/4} \qquad (14)$$

C_N is also a function of the type of enhanced tube utilized. The values of C_N for the enhanced tubes studied ranged from

$1.59N^{0.216}$ to $1.790N^{0.179}$ for an atmospheric condensing condition and from $1.321N^{.188}$ to $1.515N^{0.157}$ for a vacuum condensing condition. For smooth tubes, C_N varied from $1.07 N^{0.170}$ to $1.201_N^{0.056}$ for the atmospheric and vacuum condensing conditions.

Returning to the general empirical form

$$\frac{\overline{h_N}}{\overline{h_{Nu}}} \alpha N^{-S} \tag{15}$$

where S varies from 0.07 to 0.20, the new data above now indicates that S varies from 0.034 to 0.071 for atmospheric condensing conditions and from 0.062 to 0.093 for the vacuum condensing condition for enhanced tubes. For smooth tubes S varies from 0.08 to .194. This indicates that there is an improvement in shell side heat transfer performance using enhanced tubes over that for smooth tubes.

Catchpole and Drew [Ref. 23] utilized the three most promising tubes from their single tube experiments and evaluated them in a small multitube condenser. The condenser contained two banks of tubes each containing 15.9 mm tubes arranged in 5 staggered rows with a triangular pitch. Increasing the condensate inundation rate and increasing the concentration of noncondensable gases reduced the overall heat transfer coefficient of all tubes. However, the performance of the shaped tubes was always higher than for the smooth tubes. As an indication of the better performance, one of the shapes was always twenty-five to fifty percent better than the plain tube. Catchpole and Drew concluded that it appears that significant improvements in the overall heat transfer coefficient in steam condensers should be possible by several geometries of shaped tubes with the final choice of the tube depending upon the balance to be struck between space and weight considerations and somewhat higher operating costs, such as cleaning, increased pumping power, and retubing.

D. CONCLUSIONS

As can be seen from the brief review of prior research into the area of condensers and enhanced heat transfer techniques, the problem of condenser performance is extremely complicated. The research situation is further exacerbated by the many different experimental techniques and objectives. To completely study condenser performance, an experimental apparatus must be constructed that can investigate the effects of vapor velocity, condensate inundation and noncondensable gases either singly or in combinations.

The research shows that enhanced tubes can improve the overall heat transfer characteristics of a condenser but further understanding of how inundation mitigates this improvement must be accomplished before a final judgement can be passed.

III. EXPERIMENTAL APPARATUS

Both Eshleman and Demirel observed droplet migration in the condenser during their experimental tests. Drops moved from the tube ends toward the center and in some cases from the center to ends; this action was thought to be due to a secondary flow phenomenon in the condenser. The effect of the droplet migration is to thicken, locally, the condensate film at the center of the tube resulting in reduced heat transfer performance of the tube.

Demirel tried various modifications to the test apparatus, including venting the condenser to the atmosphere, but was unable to solve the droplet migration problem. A decision was made after Demirel's tests to redesign the test apparatus to eliminate the secondary flow problem and to make the system operate, initially, closer to the quiescent conditions of Nusselt.

The redesigned condenser used to the greatest extent possible the installed equipment from the existing rig. At the same time, a modification was made to the data acquisition system so that a simpler approach to data reduction could be utilized.

A. TEST CONDENSER

A new test section was manufactured which has a length of 305 mm and a width of 79 mm. This allows the use of a column of 16 mm O.D. active tubes flanked on each side by a column of full round dummy tubes of the same diameter. A height of 305 mm permits up to a nine tube column, but initially, only a five tube column was used.

A square inline arrangement of tubes with a spacing-to-diameter ratio of 1.5 was chosen for the initial experiments. There is, however, a provision for the dummy tubes to be rearranged to give a staggered arrangement. This was accomplished by machining slots in the end plates of the test section and providing a tube sheet for the dummy tubes which can be changed to suit the type of arrangement desired. Each end of the dummy tube is inserted into its respective tube sheet, and the tube sheets are then fitted into the end plate slots and held in place by screws.

The active tubes are located by tube sheets that are attached to the exterior of the end plates of the test section. To change the pattern of the active tubes, a new tube sheet must be constructed. Sealing of the tubes is accomplished by O-rings that are fitted into grooves machined into the tube sheets. Sealing of the tube sheet to the end plate is accomplished by gaskets.

A new diffuser was designed based upon a suggestion made by Professor T. Sarpkaya [24] of the Naval Postgraduate School. The design utilizes vanes shaped by the elastic beam equations with zero slope at the ends. The diffuser is divided by two vanes to give three chambers which approximate the natural shape of a flow leaving a nozzle. A 457 mm long, square transition piece changes the flow from a 64 mm I.D. stainless steel tube to the 79 mm by 79 mm opening of the diffuser. A vortex annihilator in the transition piece prevents vortices from entering the diffuser. The diffuser is 254 mm high with a width of 79.4 mm and a maximum length of 305 mm.

An inverted diffuser (nozzle) of the same dimensions is placed at the bottom of the test section to act as the condensate collection system and to collect the remaining steam. A 101 mm extenstion was added at the bottom of the collector which converts the system to a square 64 mm opening. To this opening, a length of 64 mm I.D. stainless steel tubing is added to which a 64 mm I.D. tube is welded at a 45 degree angle to the horizontal. The lower end of this line extends 25 mm beyond the vertical tube and acts as the condensate collection point. The condensate drains to a collection tank where it can be measured. The other end of the angled tube takes the steam to the secondary condenser.

A window was constructed so that viewing of the condensation process could be accomplished. The window, made of 203 mm by 140 mm by 13 mm pyrex plate glass, provides a maximum viewing area while satisfying structural condtions. A lip was machined in the face of the test condenser, and the glass with its frame is screwed down against this lip with a gasket to provide an air tight seal. A second piece of glass is added to the frame with an air gap between the two pieces of glass. Warm air is blown into the air gap to ensure that the glass does not fog over.

Stainless steel was utilized throughout the test section to reduce the possibilities of contamination which could lead to dropwise condensation. The exceptions are the use of aluminum for the active tube sheet and the window frames.

Figures 4-11 show the test section from various angles.

B. CONDENSATE SYSTEM

The condensate system organization, Figure 12, was unchanged with one exception; only one of the secondary condensers was utilized in an effort to reduce back pressure. However, new test section and condenser hotwells were constructed. The steam condensed in the test section is collected in the test section hotwell where it can be measured by closing an isolation valve. Opening the valve allows the condensate to feed by gravity to the secondary hotwell. Steam not condensed in the test section is condensed in the secondary condenser and then collected in its hotwell.

C. STEAM SYSTEM

The steam system schematic is shown in Figure 13. The supply is locally generated and supplied to the building housing the experimental rig. The steam is supplied via a 19 mm line and a steam inlet valve to a cast iron steam separator. From the separator, a 19 mm tube feeds the steam to two Nupro Bellows Valves to reduce the pressure and then to a desuperheater. From the desuperheater, a 64 mm I.D. stainless steel line feeds steam to the square transition piece of the test section diffuser. The steam pressure is monitored by compound gauges located between the two Nupro Valves and just before the desuperheater.

D. COOLING WATER SYSTEM

The cooling water system is a closed system as shown in Figure 14. The water is supplied from the normal house water system, passed through a water softener, and stored in a supply tank. The water is pumped from the supply tank via a 5 HP electrically driven pump. The water flows from the supply tank through a 51 mm O.D. plastic pipe to a header. Individual rotameters connected to the header control the flow through the active tubes. After leaving the rotameters, the water flows through 16 mm tubes with at least a one meter length prior to the test section to ensure fully developed flow. The cooling water is returned to the supply tank where a separate system pumps the water to a cooling tower in an effort to maintain a constant inlet temperature.

E. VACUUM SYSTEM

The mechanical vacuum pump utilized in the previous systems to maintain vacuum was replaced by an air driven air ejector. Utilizing air at a minimum of 791 kPa maintained the system at a minimum vacuum of 584 mm Hg. The air ejector pulls a suction on the secondary condenser hotwell, and it discharges through a muffler to reduce the noise hazard.

F DESUPERHEATER

A desuperheater to control sensible heat of the steam was constructed using a 318 mm diameter cylinder with a height of 508 mm. Four fan type, spray nozzles inserted equidistant around the circumference supply the cooling water. Cooling water is stored in a supply tank which may be heated electrically. A collection tank at the bottom of the desuperheater allows for drainage of the condensate. This system can be isolated via a valve as shown in the schematic, Figure 13.

G. CONDENSER TUBES

Active tubes are made of 16 mm half hard copper with 1.65 mm walls. Thermocouples are soldered into grooves within the walls of the tubes, and the high thermal conductivity ensures that the thermocouples give an accurate measurement of the wall temperature. Dummy tubes are made of stainless steel.

H. INSTRUMENTATION

1. Flow Rate

a. Cooling water flow rate is measured individually for each tube. Each flow rate was determined by a rotameter with a capacity of 70.4 LPM, and the rotameters were calibrated using the procedure listed in Appendix A of Ref. 3.

b. Steam velocity was determined by the calculation

$$\mathrm{Vel} = \frac{\dot{m}_{cond}\, v}{A_c} \tag{16}$$

where

$\dot{m}_{cond}$ = total mass flow rate of the condensate (kg/sec);

A_c = cross sectional area of the condenser (m^2); and

v = specific volume of the vapor at the test section (m^3/kg)

2. Temperature

Stainless steel sheathed copper-constantan thermocouples were utilized as the primary temperature monitoring devices. Figure 15 shows the location of the thermocouples in the condenser. Four thermocouples with a .51 mm O.D. stainless steel sheath were soldered into axial grooves machined from the center of the tube to approximately 100 mm from one end of the 600 mm long tube. The thermocouples are spaced 90 degrees apart. Two thermocouples measure cooling water inlet temperature, and four thermocouples measure cooling water outlet temperature for each tube in an attempt to get an accurate bulk temperature. Calibration of the thermocouples was accomplished using the procedures outlined in Appendix A to Ref. 3.

3. Pressure

Two different pressure sensing devices are utilized to monitor steam pressure. Two Bourdon tube gauges were located in the steam piping before the desuperheater. A mercury manometer and an absolute pressure transducer monitor the conditions at the test section. A micromanometer measures the differential pressure across the test section itself.

4. Data Collection and Display

An Autodata Nine Scanner is utilized to record and display, temperatures in degrees Celsius, obtained from the thermocouples. The signal from the pressure transducer is displayed as a voltage. Table 1 lists the channel number and location of the thermocouples.

IV EXPERIMENTAL PROCEDURES

A. OPERATING PROCEDURES

1. Preparation of Condenser Tubes

Prior to installation, each tube was properly prepared to ensure filmwise condensation. The basic cleaning procedure is outlined in Appendix A of Ref. 6.

2. System Operation and Steady State Conditions

A complete set of operating instructions is listed in Appendix A.

The parameters used to determine steady-state conditions were cooling water inlet temperature and steam saturation temperature. A steady state condition was considered achieved when conditions were reached as outlined by Eshleman [Ref. 8].

B. DATA REDUCTION PROCEDURES

To simplify the data reduction, actual tube wall temperatures were obtained. Utilizing the wall temperatures directly eliminates the necessity for the Wilson Plot technique for calculating the steam side heat transfer coefficient. The following approach using standard heat transfer equations and an energy balance was used to evaluate the raw data.

1. Total Heat Transferred per Tube

$$Q_i = \dot{m}_i \, C_i \, \Delta T_i \qquad (17)$$

where

Q_i is the total heat transferred for tube i per unit time [kJ/S or kW]

$\dot{m}_i$ is the cooling water mass flow for tube i [kg/S];

ΔT_i is the difference between outlet and inlet temperature of the cooling water for tube i [^oK]; and

C_i is the specific heat of the cooling water [kJ/kg-^oK]

2. <u>Steam Side Heat Transfer Coefficient</u>

$$h_{si} = \frac{Q_i}{A_i \ (T_s - \bar{T}_w)} \qquad (18)$$

where

h_{si} = the steam side heat transfer coefficient for tube i [W/m^2 - ^oK];

Q_i = heat transferred per unit time for tube i [W];

A_i = steam side heat transfer area [m^2];

T_s = steam saturation temperature [°K]; and

$\bar{T}_w$ = average wall surface temperature which is equal to

$$\sum_{n=1}^{4} (T_{wi})_n/4 \ [^oK] \qquad (19)$$

The steam side heat transfer coefficient is the parameter used to compare the performance of a tube in the bundle. Two assumptions were made in applying equation (18) to the data obtained from the experimental apparatus.

a. Resistance due to noncondensable gases was negligible. The system was tested for tightness and was found to be satisfactory. Noncondensable gases brought in by the steam (if any) were exhausted by the air ejector. Good steam flow conditions in the test section continually swept noncondensable gases away from the test tubes.

b. The resistance of the half hard copper tube was also assumed negligible. It was assumed that the thermocouples recorded actual tube wall surface temperature.

C. DATA REDUCTION PROGRAM

A computer program to analyze the data was constructed in Basic Language for use on an HP-85 Computer System. A peripheral plotter is utilized to draw the plots required for analysis of bundle performance. The computer program is listed in Appendix D.

V. RESULTS AND DISCUSSION

Due to unavoidable delays in constructing the new test condenser, there was insufficient time to do inundation research. However, initial tests with steam to validate system operation were performed.

Observation of the condensate droplet formation showed that there was no droplet migration longitudinally. Therefore, it is felt that there are no secondary flow characteristics in the new test section itself. This is attributed, primarily, to the new diffuser design as well as to the decrease in length of the condenser.

Additional results from the initial tests indicate that the air ejector will provide the required vacuum. At the present time, air leaks at joints and welds limit the vacuum in the system to 508 mm of Hg. The single secondary condenser is adequately sized to handle the maximum steam mass flow rate available from the house steam supply.

Several design problems surfaced during the initial operation. First, the tube wall thermocouples are very fragile and are easily broken. This is especially true where the thermocouple leaves its groove. A better method must be developed to handle installation and securing of the thermocouples.

Secondly, cleaning of the tubes especially after installation is difficult. This is due to the smaller size of the condenser and the lack of access to its interior. Since the dummy tubes do not come out easily, there is not enough space to get a brush with the cleaning fluid to the center (active) tubes. A redesign of the method of attaching the dummy tube sheets to the ends of the condenser would eliminate this problem.

Finally, the spacing-to-diameter ration of 1.5 limits the means of sealing required for the active tubes passing through the aluminum tube sheets. O-rings were the only practical means of providing the necessary sealing, but they require close tolerances for good performance. The close tolerances means that a great deal of force is required to push the tube through the tube sheet at the risk of damaging the fragile tube wall thermocouples. Additionally, the close tolerances reduce the frequency that the active tubes can be removed conveniently for cleaning. Expanding the spacing-to-diamter ratio, however, reduces the applicability to actual condensers. A more subtle method of tube sealing must be found so that this problem can be more satisfactorily resolved.

VI. CONCLUSION AND RECOMMENDATIONS

After testing of the new test condenser, it was concluded that the new apparatus will satisfactorily produce data on inundation effects with no droplet migration. This satisfies thesis objective number one.

The following suggestions are submitted for the purpose of improving the test condenser:

1. Revise the cleaning process of the test condenser especially when the condenser is completely built and the tubes are in place.

2. Redesign the method of mounting the fragile tube wall thermocouples so that they will not be damaged during handling and installation.

3. Redesign the active tube sheet so that the active tubes can be easily inserted and removed.

The following recommendations for future work are provided:

1. Continue design development work to ensure continued high performance of the apparatus.

2. A comparison of the experimental steam side heat transfer coefficient versus the Nusselt theoretical value is required to ensure the validity of the rest results.

3. Obtain a comparison of the accuracy of the Wilson Plot technique versus the approach developed in this work for the steam side heat transfer coefficient.

4. Obtain baseline data using smooth tubes (thesis objective number two).

5. Obtain data utilizing enhanced tubes.

TABLES

TABLE I

Channel Numbers for Stainless Steel Sheathed Copper Constantan Thermocouples.

Location	Channel	Location	Channel	Location	Channel
T_s	67	T_{co} #3	87	T_w #1	43
T_s	68	T_{co} #3	88	T_w #2	44
T_s	69	T_{co} #3	89	T_w #2	45
T_{ci} #1	90	T_{ci} #4	74	T_w #2	46
T_{ci} #1	91	T_{co} #4	75	T_w #2	47
T_{co} #1	96	T_{co} #4	84	T_w #3	48
T_{co} #1	97	T_{co} #4	85	T_w #3	49
T_{co} #1	98	T_{co} #4	85	T_w #3	50
T_{co} #1	99	T_{ci} #5	80	T_w #3	51
T_{ci} #2	70	T_{ci} #5	81	T_w #4	52
T_{ci} #2	71	T_{co} #5	76	T_w #4	53
T_{co} #2	82	T_{co} #5	77	T_w #4	54
T_{co} #2	93	T_{co} #5	78	T_w #4	55
T_{co} #2	94	T_{co} #5	79	T_w #5	56
T_{co} #2	95	T_w #1	40	T_w #5	57
T_{ci} #3	72	T_w #1	41	T_w #5	58
T_{ci} #3	73	T_w #1	42	T_w #5	59
T_{co} #3	86				

FIGURES

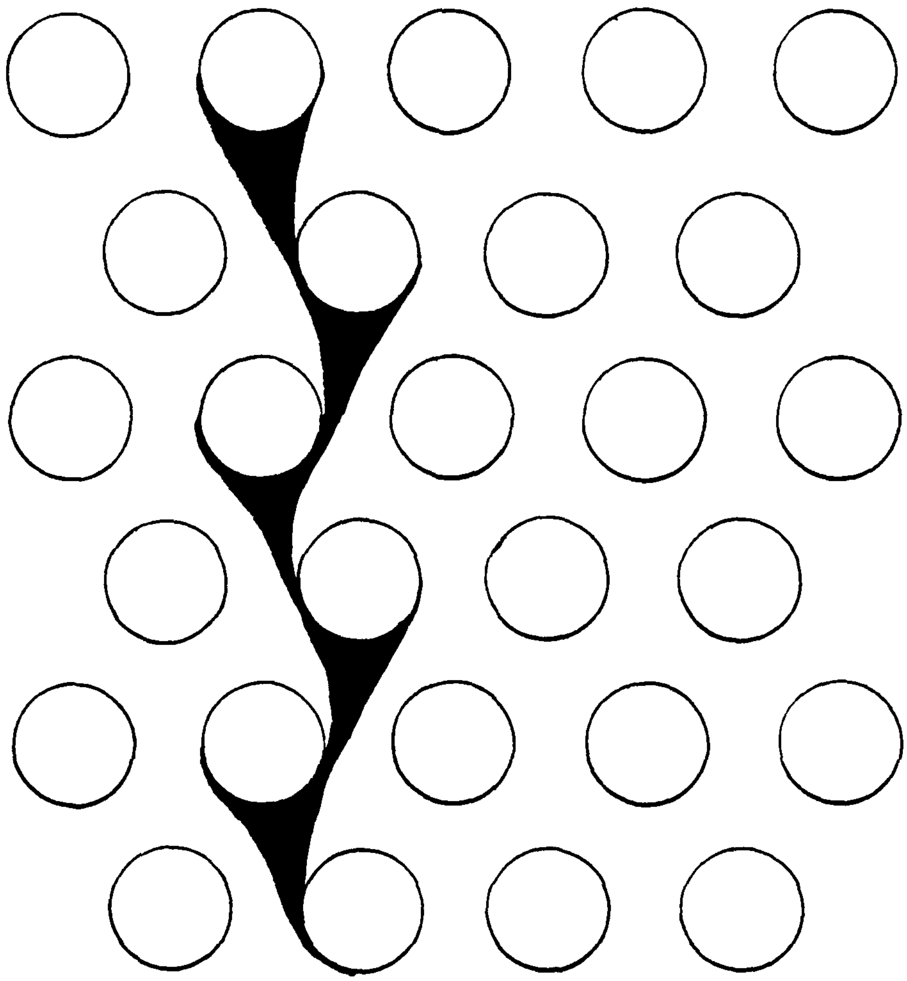

Figure 1: Droplet Path Through a Tube Bundle with Side Drainage

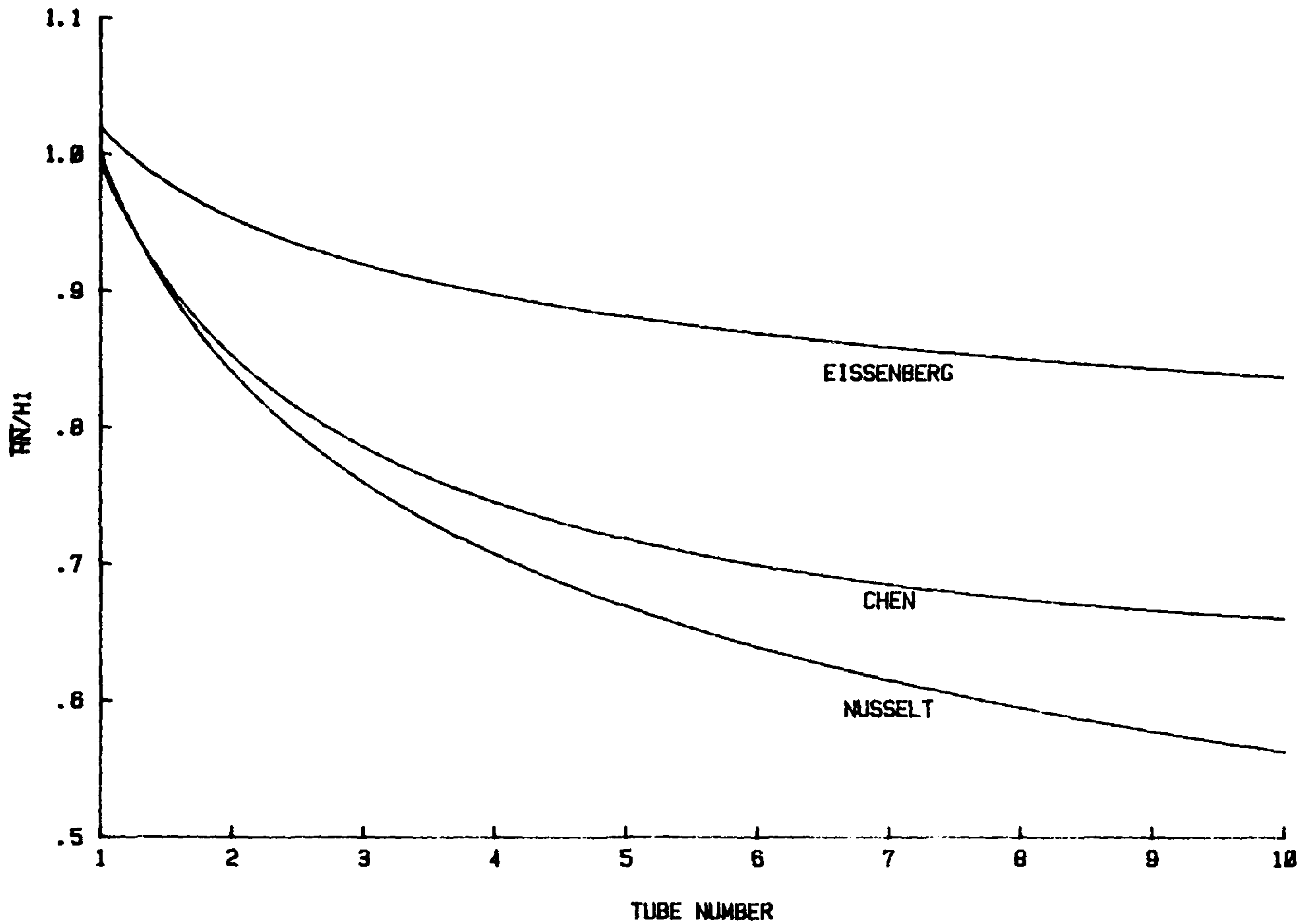

FIG. 2 COMPARISON OF PREDICTIONS OF TUBE BUNDLE PERFORMANCE

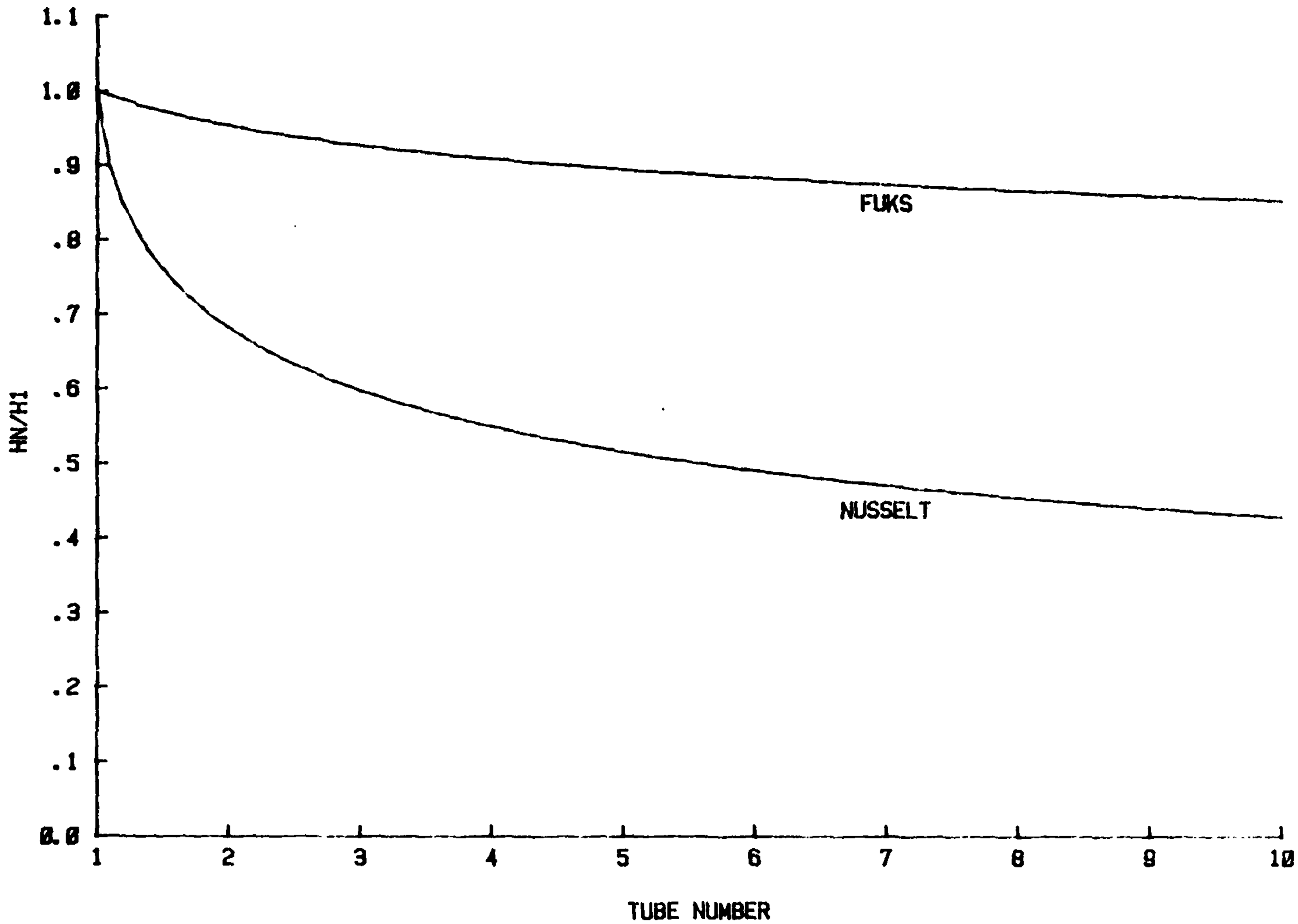

FIG. 3 COMPARISON OF PREDICTIONS FOR THE LOCAL HEAT TRANSFER COEFFICIENT

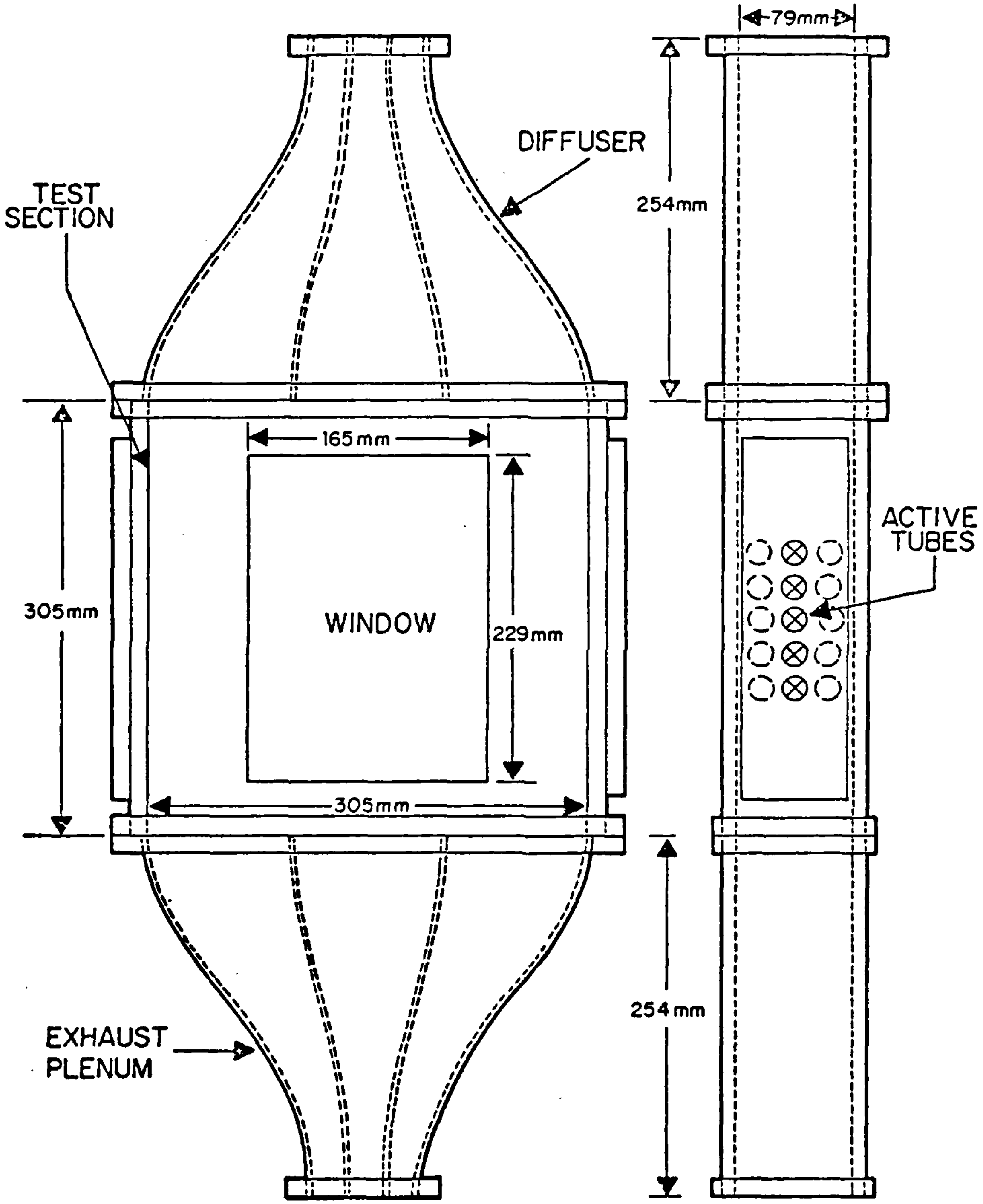

Figure 4. **Sketch of Test** Condenser

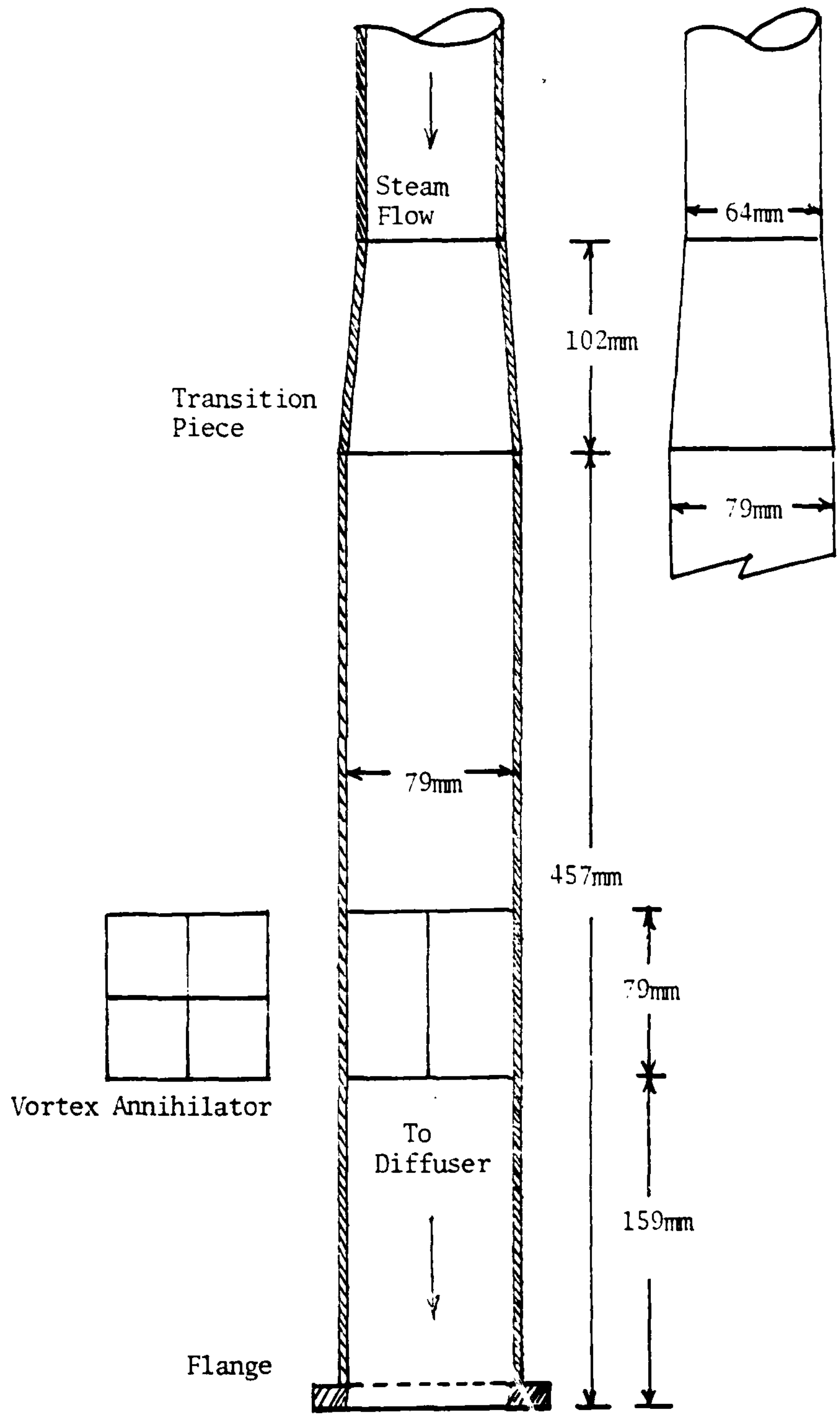

Figure 5. Details of Transition Piece and Vortex Annihilator

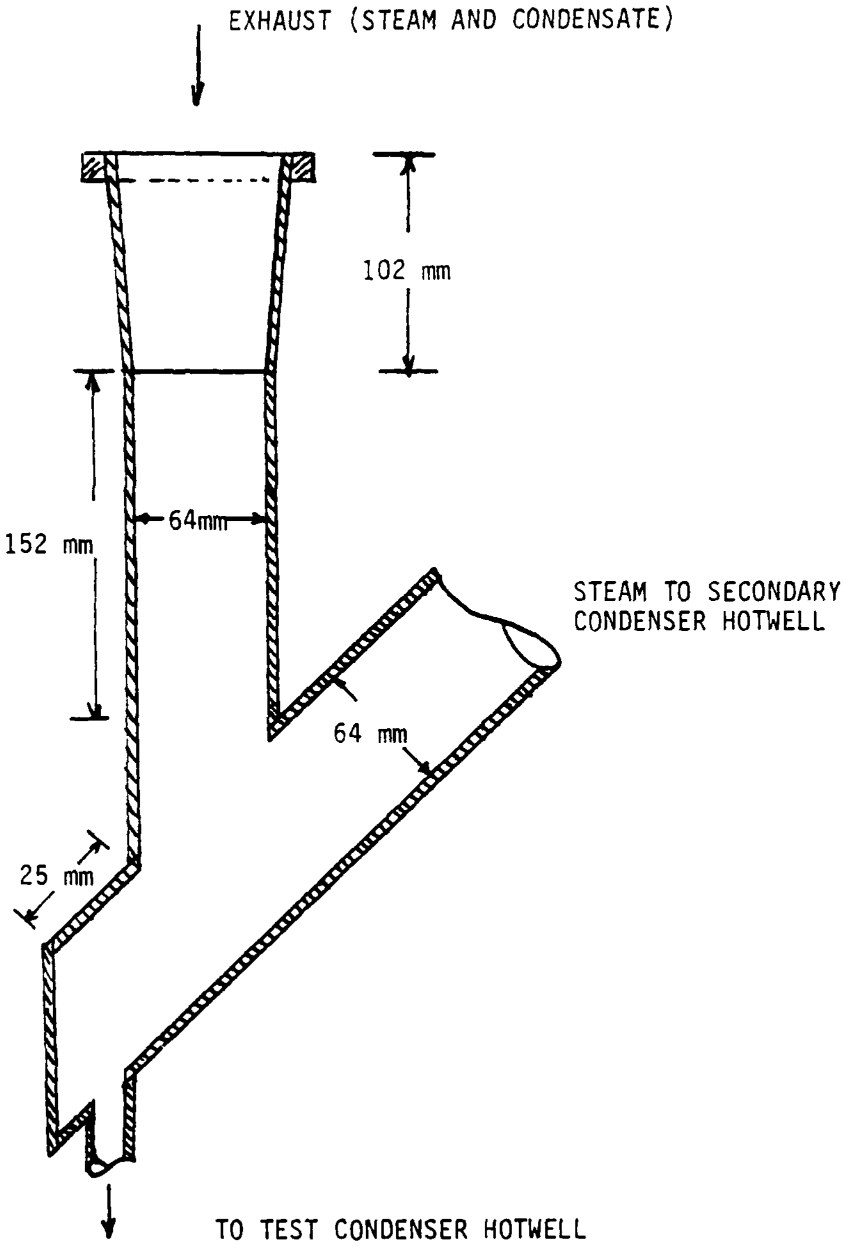

Figure 6. Details of Exhaust and Condensate Piping from the Exhaust Plenum

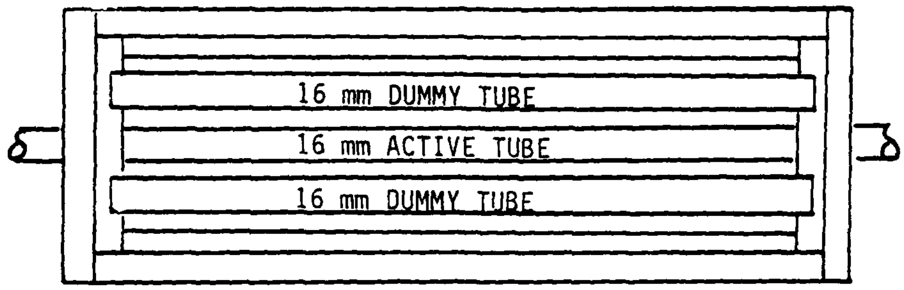

Figure 7. Top View of Test Section

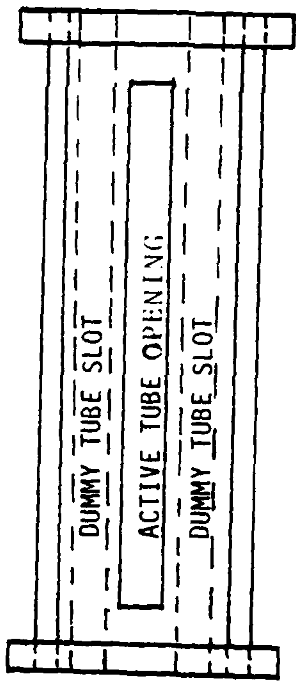

Figure 8. Side View of Test Section

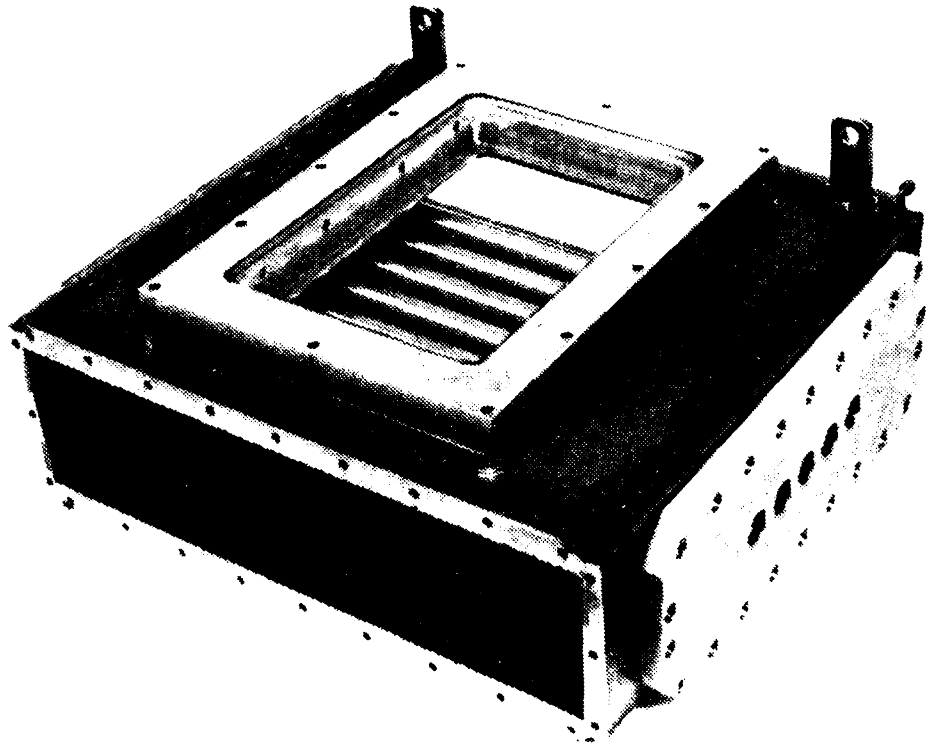

Figure 9. Photograph of Test Section

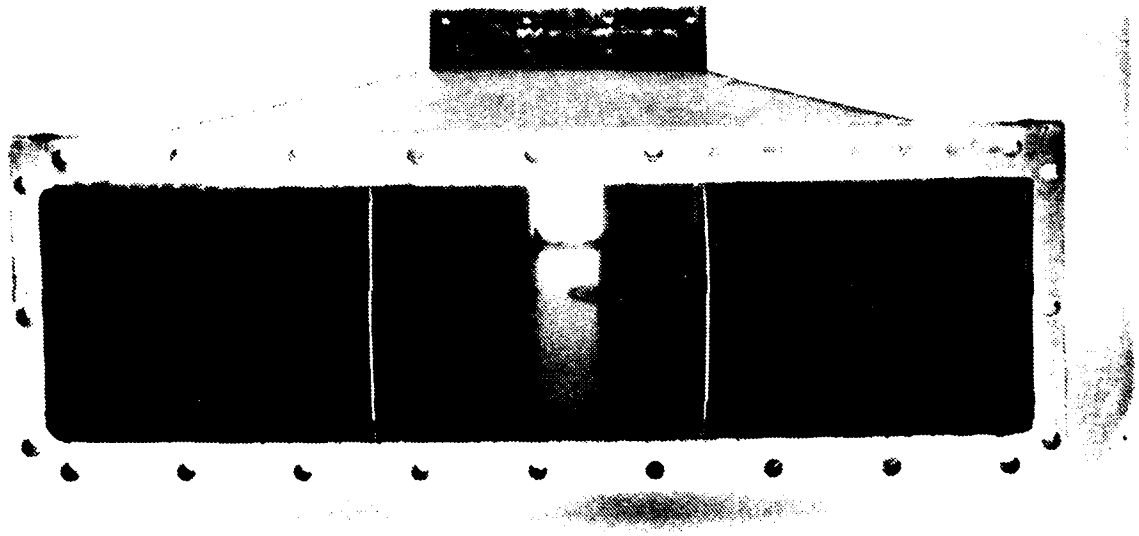

Figure 10. Photograph of Diffuser Showing Vanes

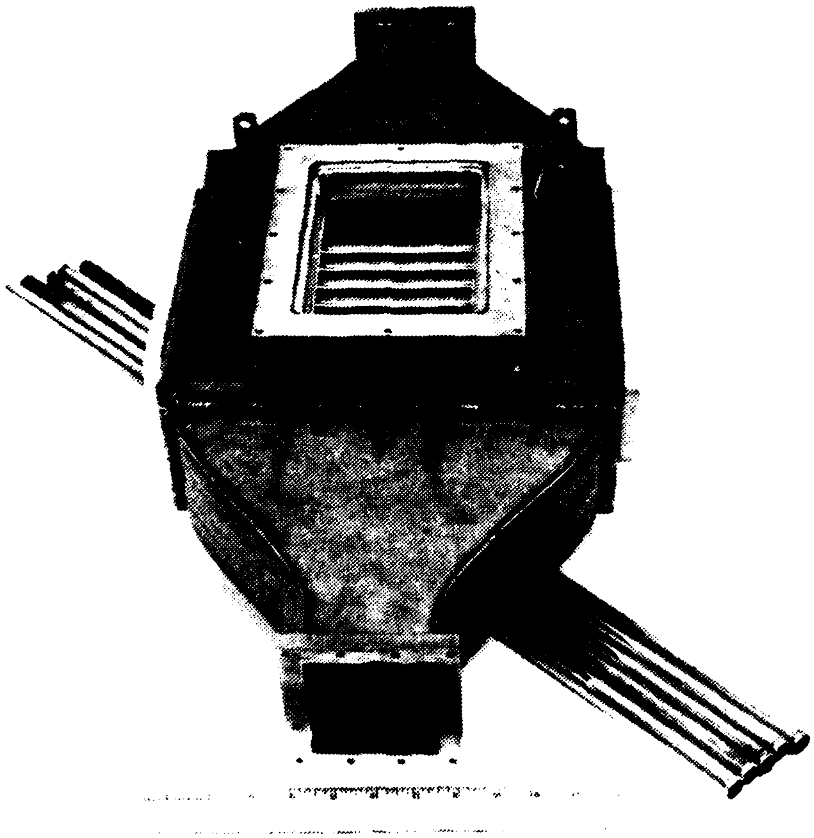

Figure 11. Photograph of Test Condenser

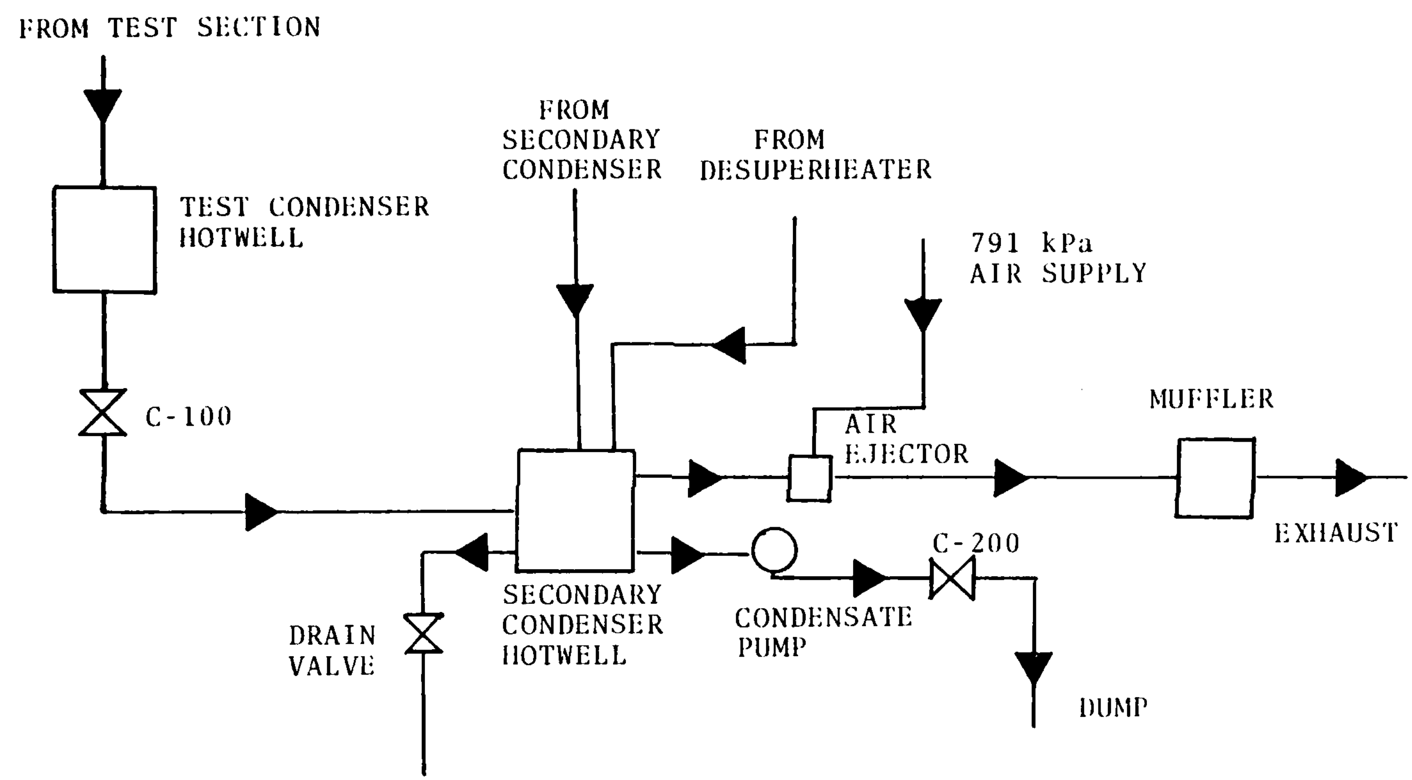

Figure 12. Schematic Diagram of Condensate and Vacuum Systems

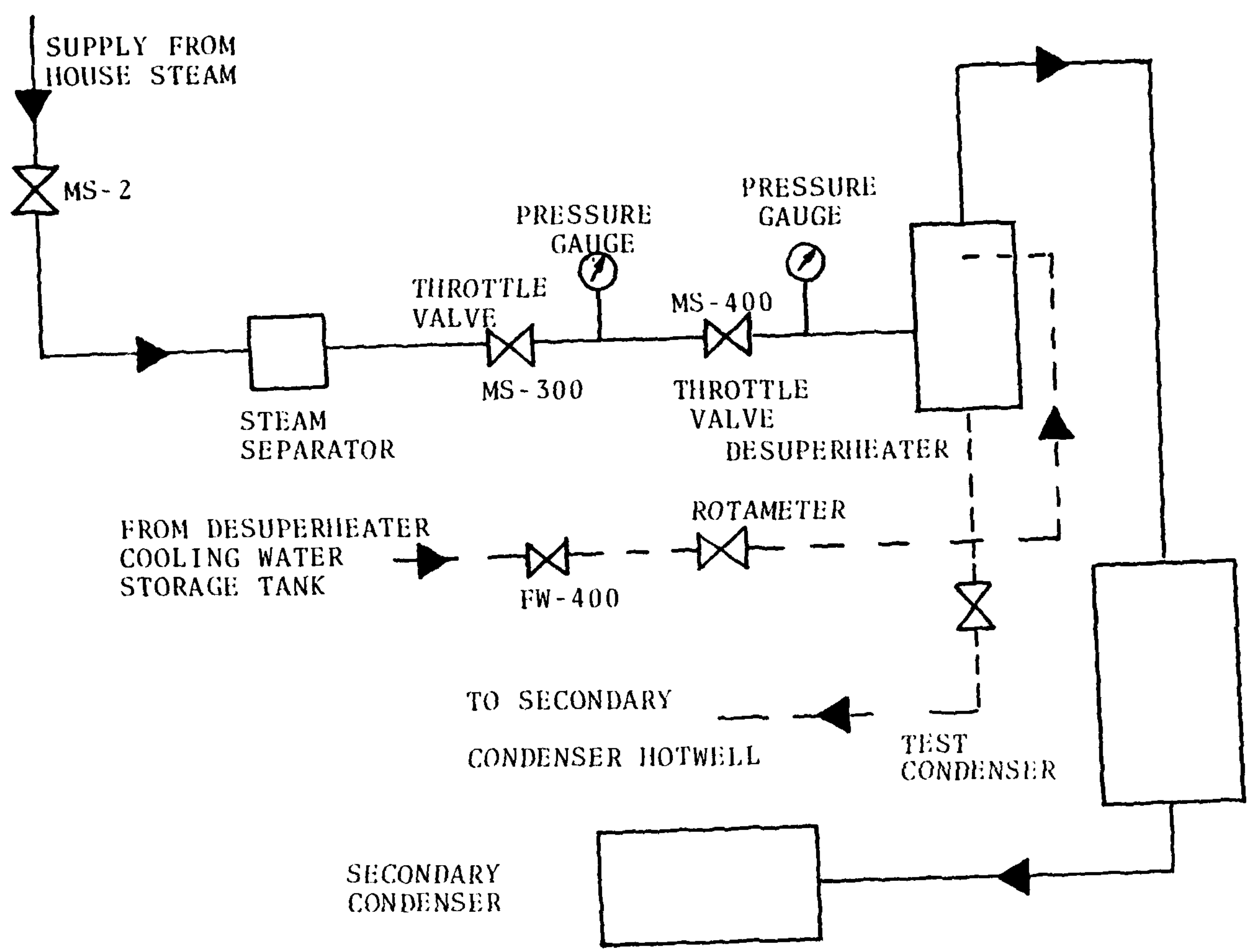

Figure 13. Schematic Diagram of Steam System

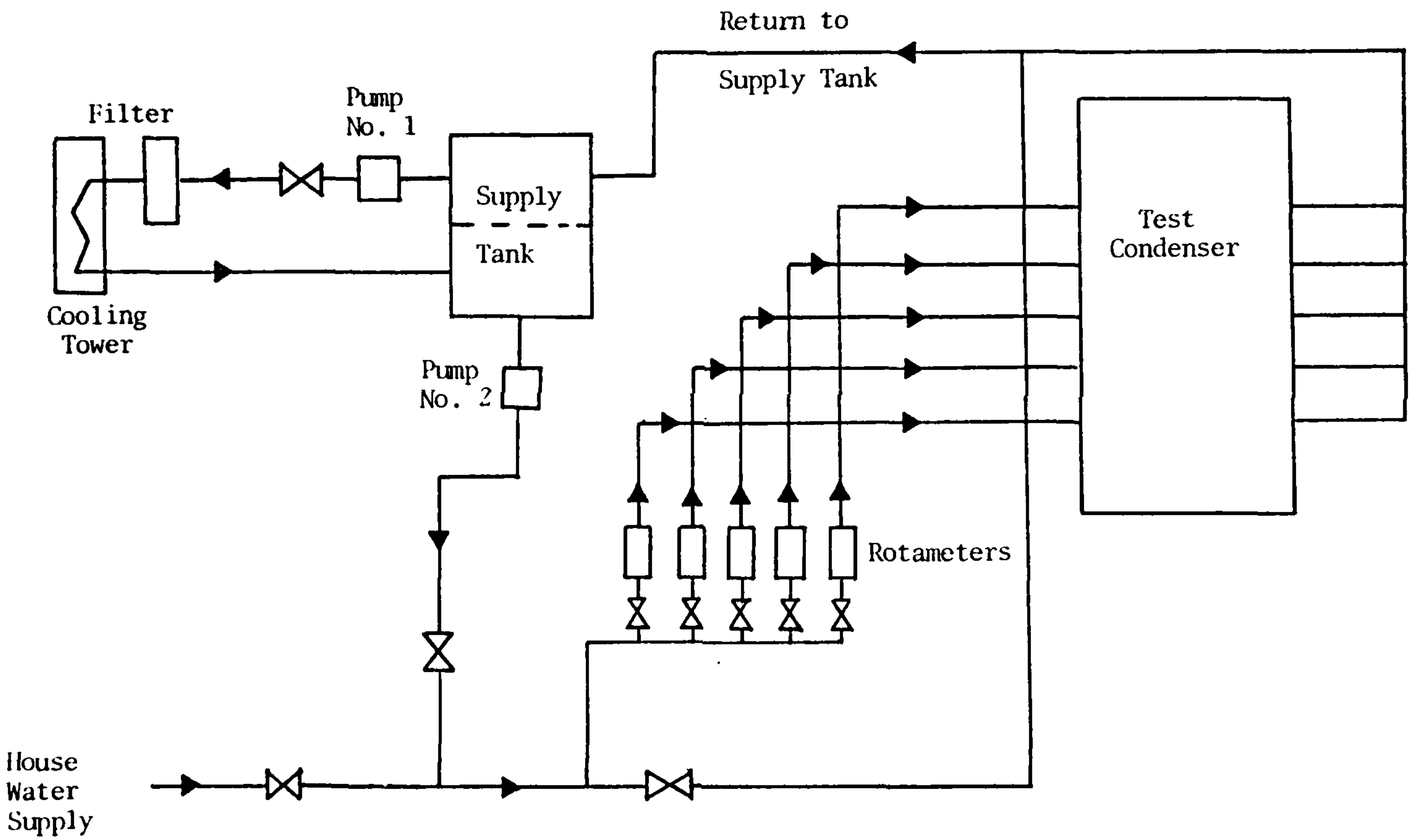

Figure 14. Schematic Diagram of Cooling Water System.

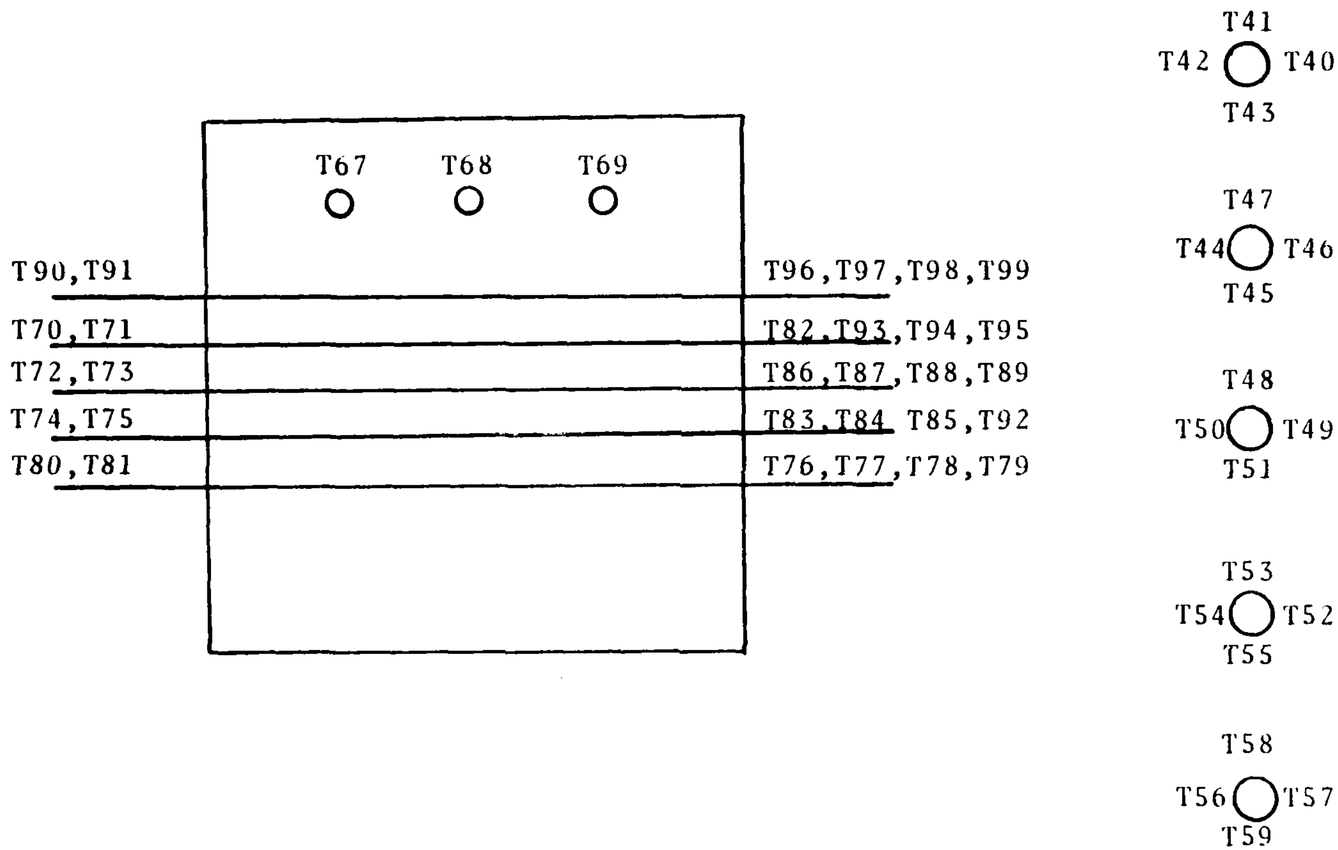

Figure 15. Location of Thermocouples

APPENDIX A: OPERATING PROCEDURES

1. Initial Procedures

(a) Energize main circuit breaker located in Power Panel P-2;

(b) Turn key switch on (located on right side of main control board);

(c) Energize circuit breaker on the left side of main control panel by pressing start button;

(d) Energize individual circuit breakers on left side of main control panel. The following list identifies each circuit breaker:

1. #1 - Feed pump
2. #2 - Outlets
3. #3 - Hot water heater (feedwater tank) (not required)
4. #4 - Condensate pumps
5. #5 - Boiler (not required)
6. #6 - Cooling tower
7. #7 - Cooling water pump;

(e) Energize instrumentation (see Appendix B);

(f) Open valve to FW-400 to desuperheater cooling water rotameter (set rotameter to 15-25% flow); and

(g) Ensure that 791 kPa air is supplied to air ejector and that a minimum vacuum of 508 mm Hg is maintained at the secondary condenser hotwell.

2. Operation

(a) Cooling Water System

1. Open valves CW-1 and CW-2 two turns;
2. Open valve CW-4, then energize pump No. 1 and pump No. 2;
3. Open valve CW-2, and open valve CW-1 so that flowmeter is set at 40-50% flow;
4. Open valves CW-5, CW-6, CW-7, CW-8, and CW-9 to obtain desired cooling water rate;
5. Adjust valve CW-4 as necessary to maintain desired flow rate;
6. Vent both sides of the manometer; and
7. Open valve DS-7 to start flow to secondary condenser.

(b) House Steam System

1. Ensure that the throttle valves MS-300 and MS-400 are closed;
2. Open valve MS-2 fully;
3. Open valve MS-300 fully; and
4. Open valve MS-400 until desired pressure is obtained on the pressure gage.

(c) Condensate System

1. To collect drains in the test condenser hotwell, operate the system with valve C-1-- closed. After the test run has been completed, open valve C-100,

and the condensate will drain into the secondary condenser hotwell; and

2. The condensate pump is operated intermittently. When the level in the secondary condenser dictates, start the condensate pump and open valve C-200. When the condensate level has droped a sufficient amount, secure valve C-200 and then secure the condesate pump.

3. Securing the System

(a) Secure valve MS-2;

(b) Secure valve MS-300;

(c) Secure valve MS-400;

(d) Open valve C-100;

(e) Pump the condensate from the secondary condenser hotwell. When this is complete, close valve C-200, secure the condensate pump, and close valve C-100;

(f) Secure the air to the air ejector;

(g) Secure flow to the secondary condenser;

(h) Secure pump No. 1 and pump No. 2 and close valve CW-5, CW-6, CW-7, CW-8 and CW-9;

(i) Secure instrumentation;

(j) De-energize by securing circuit breakers at the main control panel; and

(k) De-energize at the main circuit breaker P-2.

4. Secondary Systems

(a) Vacuum System

The vacuum in the steam system is controlled by the air ejector. A 100 psig air supply is the motive force for the air ejector which takes a suction on the secondary condenser hotwell.

(b) Desuperheater

Valve FW-400 controls the flow of cooling water to the desuperheater spray nozzles. Optimum flow level is between 20 and 30 percent flow on the rotameter.

5. Safety Devices

To secure all power to the system in an emergency, depress the red button on the right of the main control panel next to the key switch.

APPENDIX B: AUTODATA NINE SCANNER OPERATION

1. Energize Instrumentation

(a) Set Time

1. all alarms and output switches off;
2. set date/time on thumbwheels;
3. set the display switch to "time"; and
4. lift "set time" to switch.

(b) Assigning Multiple Channels

1. set display switch to "all";
2. check that all alarms and switches are "off";
3. set the scan switch to continuous;
4. lift the "slow" switch;
5. set the first channel thumbwheels to "000" and last channel thumbwheels to "001;
6. to assign channel "0" and "1" depress and hold the 10V and HI RES buttons for at least one scan and lift scan start switch to start scanning;
7. set the last channel thumbwheels to "039" before setting the first channel thumbwheels to "001";
8. depress the skip button and lift the scan switch to skip channels 001 thru 039;
9. set the last channel thumbwheels to "099" before setting the first channel thumbwheels to "040"; and

10. to assign channels depress and hold the T/^{o}C and HI RES buttons for at least one complete scan.

(c) Interval Scan

1. set the thumbwheels to the interval desired between scans;
2. depress the "stop.enter" switch;
3. set the display switch to "interval";
4. depress the "set interval" switch;
5. set the scan switch to "interval";
6. set the first channel thumbwheels to "040";
7. set the last channel thumbwheels to "099 "; and
8. lift the "scan start" switch.

(d) Optional Devices as Required

1. printer on/off
2. "slow" switch
3. single channel display.

APPENDIX: C

ERROR ANALYSIS

The general form of the Kline and McClintock [25] "second order" equation is used to compute the probable error in the results. For some resultant, R, which is a function of primary variables x_1, x_2, x_3, ..., x_n, the probable error in R, δR, is given by:

$$\delta R = [(\frac{\delta R}{\delta X_1} \delta X_1)^2 + (\frac{\delta R}{\delta X_2} \delta X_2)^2 + \ldots + (\frac{\delta R}{\delta X_n} \delta X_n)^2]^{\frac{1}{2}}$$

where δX_1, δX_2, ..., δX_n is the probable error in each of the measured variables.

1. Uncertainty in the Heat Transferred, Q

$$Q = \dot{m}\ Cp\ (T_{out} - T_{in})$$

$$\frac{\delta Q}{Q} = [(\frac{\delta \dot{m}}{\dot{m}})^2 + (\frac{\delta Cp}{Cp})^2 + (\frac{\delta T_{out}}{T_{out} - T_{in}})^2 + (\frac{\delta T_{in}}{T_{out} - T_{in}})^2]^{\frac{1}{2}}$$

2. Uncertainty Analysis of the Steam Side Heat Transfer Coefficient

$$h_s = \frac{Q}{A\ (T_{sv} - \overline{T}_w)}$$

$$\frac{\delta h_s}{h_s} = [(\frac{\delta Q}{Q})^2 + (\frac{\delta A}{A})^2 + (\frac{\delta T_{sv}}{T_{sv} - \overline{T}_w})^2 + (\frac{\delta \overline{T}_w}{T_{sv} - \overline{T}_w})^2]^{\frac{1}{2}}$$

APPENDIX D: COMPUTER PROGRAM AND DOCUMENTATION

The computer program listed in Table D-1 uses the HP-85 Computing System and the Basic Language. The program uses the inter-active approach where directions or questions are displayed on the cathode ray tube screen to which the operator must respond.

1. Directions/Questions of the Computer Program

The following is a list of the directions/questions and the responses required by the operator.

(a) Line 50 - Enter run number.

The operator inserts the designator of the experimental run.

(b) Line 80 - What is the number of tubes?

The operator inserts a number from 1-9 corresponding to the number of active tubes used in the experiment.

(c) Line 130 - Data input.

The operator provides the following information.

1. The two cooling water inlet temperatures in ^{o}C.
2. The four cooling water outlet temperatures in ^{o}C.

3. The three steam saturation temperatures in ^{o}C.

(d) Line 150 - Data input

The operator provides the four tube wall temperatures in ^{o}C.

(e) Line 170 - Data input

The operator provides;

1. the cooling water mass flow rate in kG/min;

2. the specific heat of the cooling water in kJ/kg·K.

Items C - E will be displayed for each active tube.

2. <u>Computational Procedures of the Computer Program</u>

There are several computational sections in the program.

(a) Lines 210 - 240 - <u>Average Temperature Calculations</u>.

The average inlet, outlet, and steam saturation temperatures are calculated.

(b) Line 310 - Heat transferred.

The heat transferred by the water is calculated using the equation:

$$Q = (\dot{m}C \quad \Delta T)\ \text{water} \qquad \text{(D-1)}$$

(c) Line 330 - The outside heat transfer coefficient.

The outside heat transfer coefficient is calculated from:

$$h_s = \frac{Q}{(A_{tube})\ (T_s - T_w)}$$

(d) Lines 340-350 - Calculation of the average heat transfer coefficient.

(e) Line 360 - Determination of the normalized heat transfer coefficient.
The normalized heat transfer coefficient is found by dividing the average heat transfer coefficient of the first tube.

3. Information Output

The following information is printed by the computer program.

(a) Line 250 - Average outlet temperature in ^{o}C

(b) Line 260 - Average inlet temperature in ^{o}C

(c) Line 270 - Average wall temperature in ^{o}C

(d) Line 280 - Average steam saturation temperature in ^{o}C

(e) Line 320 - Tube number

(f) Line 370 - Normalized heat transfer coefficient

4. Graphics Output

The graph of the normalized heat transfer coefficient versus the tube number is plotted using the peripheral printer. The operator must ensure that a blank 8½" x 11" piece of paper is in the printer, and to continue the program the "continue" button must be pressed.

TABLE D-1

```
10   ! RHM5 plots HN/H1 vs tube number
20   Option base 1
30   DIM Q(9), M(9), I1(9), 12(9), 01(9), 02(9), 03(9), 04(9),
     I(9), 0(9), K(9), T1(9), T2(9), T3(9)
40   DIM H(9), T0(9), T4(9), T5(9), T6(9), T7(9), T(9)
50   DISP "Enter run number"
60   Input V
70   Print "Run number is", V
80   DISP "What is the number of tubes"
90   Input J
100  A1 = PI
110  K1 = 0
120  For N = 1 to J
130  DISP "Enter the cooling water inlet and outlet temperature
     and the steam saturation temperature"
140  Input I1(N), I2(N), 01(N), 02(N), 03(N), 04(N), T1(N),
     T2(N), T3(N),
150  DISP "Enter the tube wall temperatures"
160  Input T4(N), T5(N), T6(N), T7(N)
170  DISP "Enter the cooling water mass flow rate and specific
     heat"
180  Input M(N),C(N)
190  Next N
200  For N = 1 to J
210  0(N) = (01(N) + 02(N) + 03(N) +04(N))/4
220  I(N) = (I1(N) + I2(N))/2
230  T(N) = (T4(N) + T5(N) + T6(N) + T7(N))/4
240  T0(N) = (T1(N) + T2(N) + T3(N))/3
250  Print "0 =", 0(N)
260  Print "I =", I(N)
270  Print "T =", T(N)
280  Print "T0 =", T0(N)
290  Next N
300  For N = 1 to J
310  Q(N) = M(N)x C(N) x 0(N) - I(N))/60
320  Print "Tube number =", N
330  K(N) = Q(N)/A1/(T0(N) - T(N))
340  K1 = K1 + K(N)
350  K2 = K1/N
360  H(N) = K2/K(1)
370  Print "N.H.T.C. =", H(N)
380  Print "Q =", Q(N)
390  DISP "Ensure paper is in the plotter"
400  Pause
410  Plotter is 705
```

```
420   Limit 32, 235, 28, 182
430   Locate 10, 130, 15, 96
440   Scale 1, 10, 5, 1.1
450   FXD 0, 1
460   LAXES - 1, .1, 1, .5, 1, .1
470   SETGU
480   Move 71, 7.5
490   DEG @ LDIR 0
500   LORG 5
510   Label "Tube Number"
520   Move 3, 56.5
530   DEG @ LDIR 90
540   LORG 5
550   Label "HN/H1
560   Move 1, 55
570   LDIR 90
580   LORG 5
590   Label "--"
600   Move 71, 2.5
610   LDIR 0
620   LORG 5
630   Label "Fig.   HN/H1 vs tube number run #", V
640   Move 58, 5, 4.5
650   LDIR 0
660   LORG 5
670   Label "--"
680   SETUU
690   Move 1,
700   For T = 1 to 10 step .1
710   R = T^-.25
720   Draw T, R
730   Next T
740   Move 1, 1
750   For T = 1 to 10 step .1
760   R = 6 + .42 x T^-.25
770   Draw T, R
780   Next T
790   For N = 1 to J
800   Move N, H(N)
810   Label "x"
820   Next N
830   END
```

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